GARRETT

AIRESEARCH MANUFACTURING COMPANY

INVESTIGATION OF COMPLIANT
FOIL BEARINGS FOR LONG-LIFE
LIQUID HYDROGEN PUMPS

PHASE II BEARING TESTS

CONTRACT NAS 1-15807

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Revision	Date	Pages Affected (Revised, Added, Eliminated)

1. INTRODUCTION

In Phase I of the program, the feasibility of using a foil bearing in an LH2 pump was analyzed. Figure 1-1 shows the preliminary pump design used in that analysis. This pump consists of a two-stage impeller radially supported by a foil journal bearing. Axial thrust loads are supported outside of the pump section by an angular contact ball bearing.

It is concluded in the Phase I report that foil bearings are suitable for LH_2 pumps, and that a journal foil bearing diameter of 1.75 to 2.00 in. with a length-to-diameter ratio of 1.0 to 1.5 is adequate for the preliminary pump design.

During Phase II bench tests were run to verify the feasibility study conducted in Phase I. The original Statement of Work called for direct bearing tests in an LH2 environment; however, after studying the test facilities and equipment available at AiResearch, it was agreed to reduce the scope of the test program to stay within budget.

In the revised work statement, feasibility is to be verified by testing an existing foil journal bearing in water under conditions hydrodynamically similar to those in an LH_2 pump. An existing AiResearch bearing of 1.75 in. dia by 2.4 in. long was selected, and all intended tests were successfully completed. Important bearing performance parameters, including load capacity, running torque, shaft dynamics, and wear were monitored. The feasibility of using foil bearings in LH_2 pumps is thus verified.

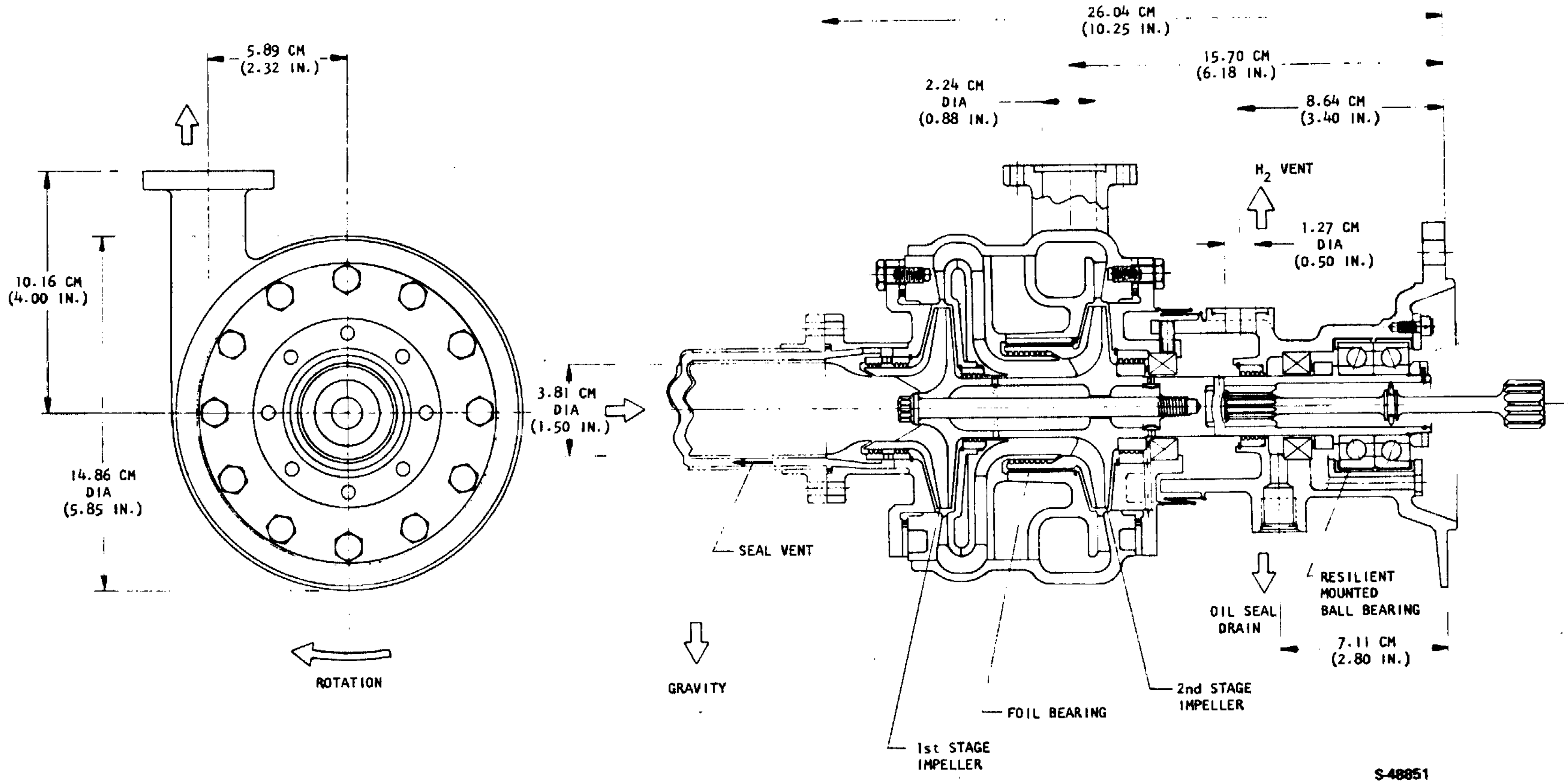

Figure 1-1. Engine High-Pressure LH_2 Pump

2. DESCRIPTION OF TEST RIG

An existing air bearing test rig was modified to run distilled water instead of air. The modified test rig consisted of a shaft supported by a ball bearing on one end, with the test bearing on the other (Figure 2-1). A radial load was applied through a ball bearing that was mounted between the shaft support bearings by a pneumatic piston. The shaft was driven through a flexible coupling by a variable-speed electric motor. Shaft speed was measured through a fiber optic probe from a 30-tooth gear mounted on the shaft.

The test bearing was mounted in a dynamometer supported by an air hydrostatic bearing so that the bearing running and starting torques could be accurately measured. Labyrinth seals were used to separate air and water and to control water leakage. An eddy current proximity probe was installed near the test bearing (see Figure 2-1) to monitor the shaft dynamics.

The fluid supply schematic is shown in Figure 2-2. The system consisted of a 50-gal water storage tank with a supply pump. Water at 72°F was supplied at about 18 psig throughout the tests to ensure a flooded condition in the bearing cavity.

Figure 2-3 is a picture of the dynamometer end of the test rig. The overall test setup is shown in Figure 2-4.

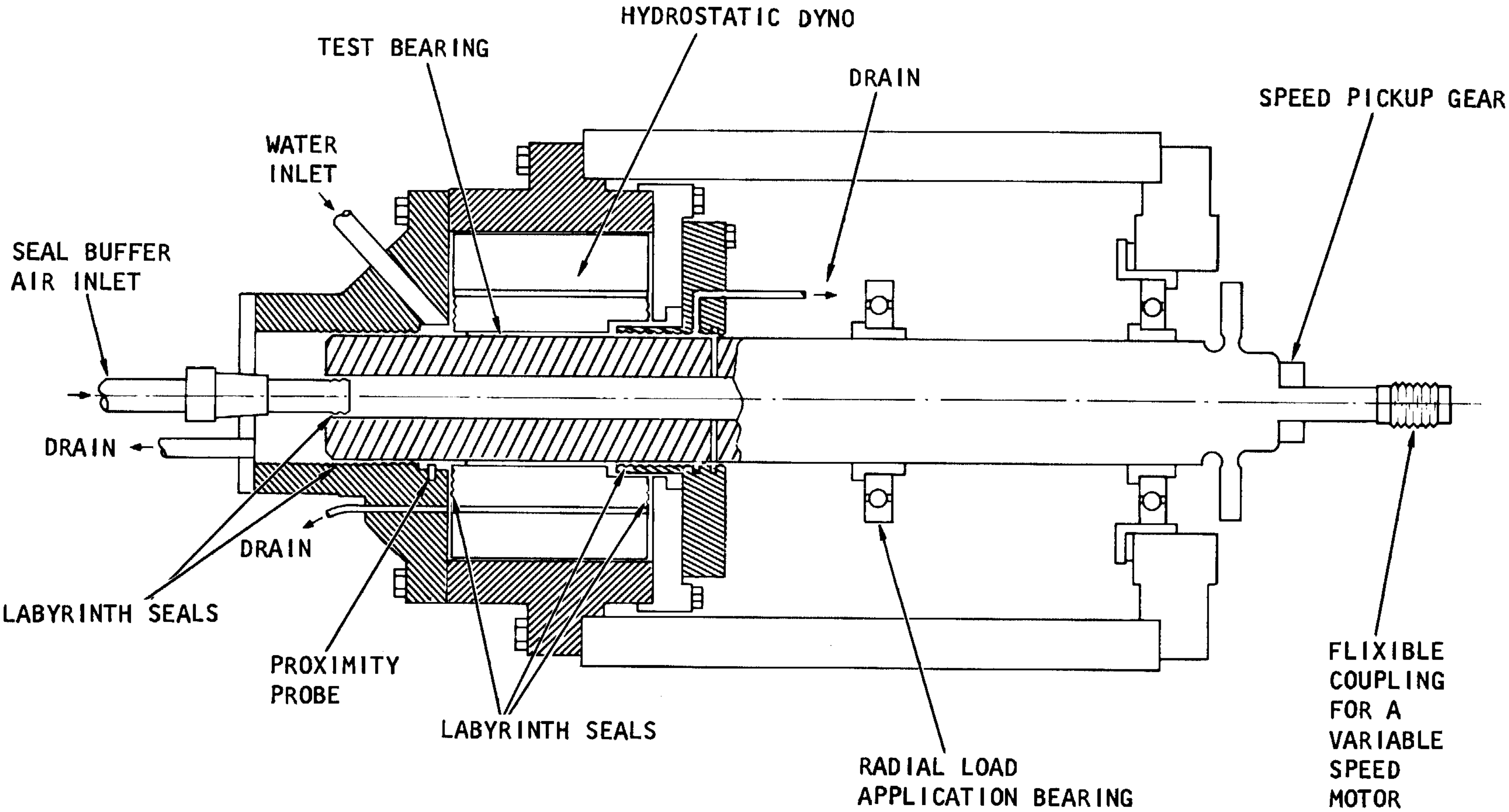

Figure 2-1. Test Rig Schematic

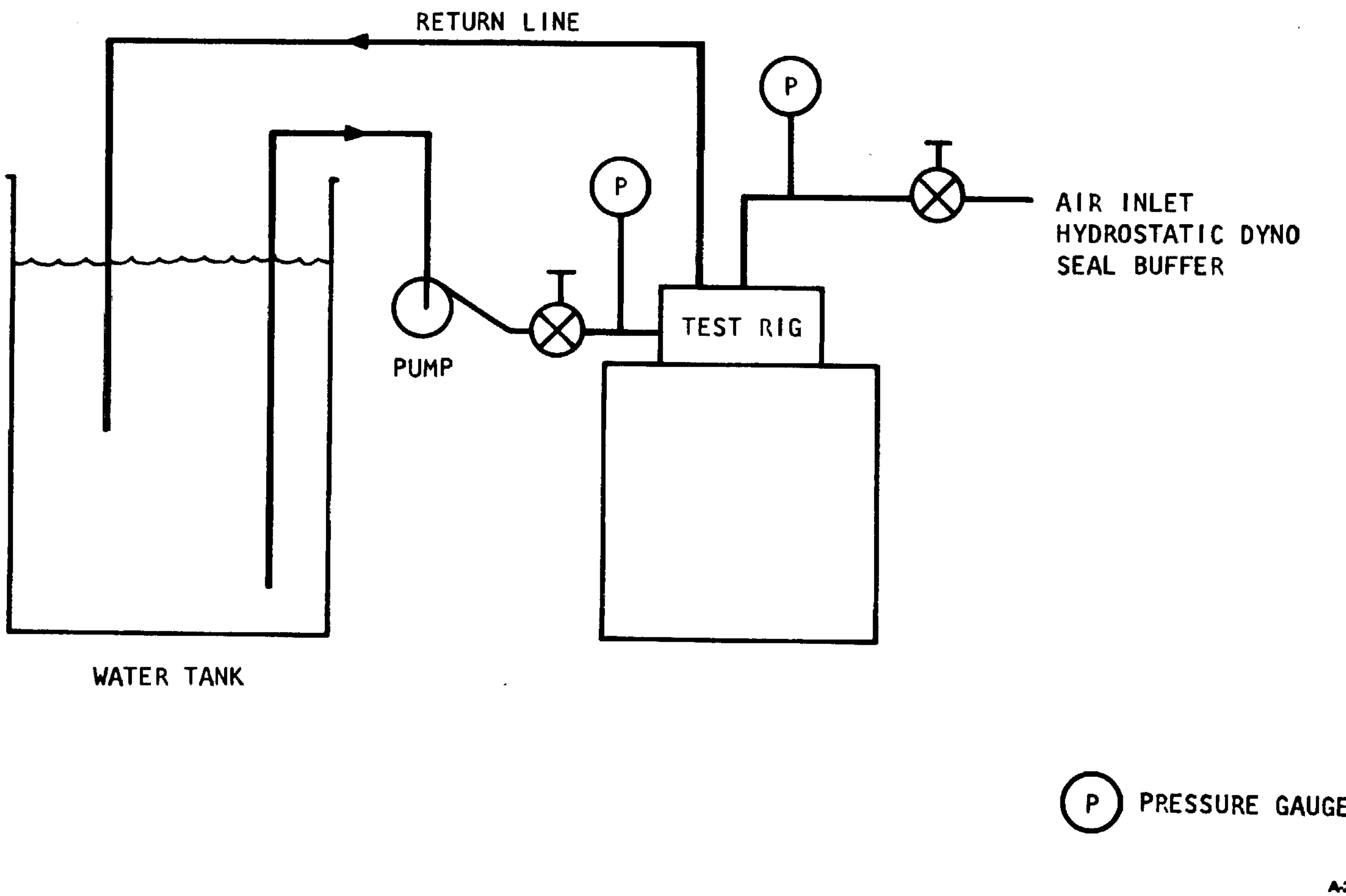

Figure 2-2. Fluid Supply Schematic

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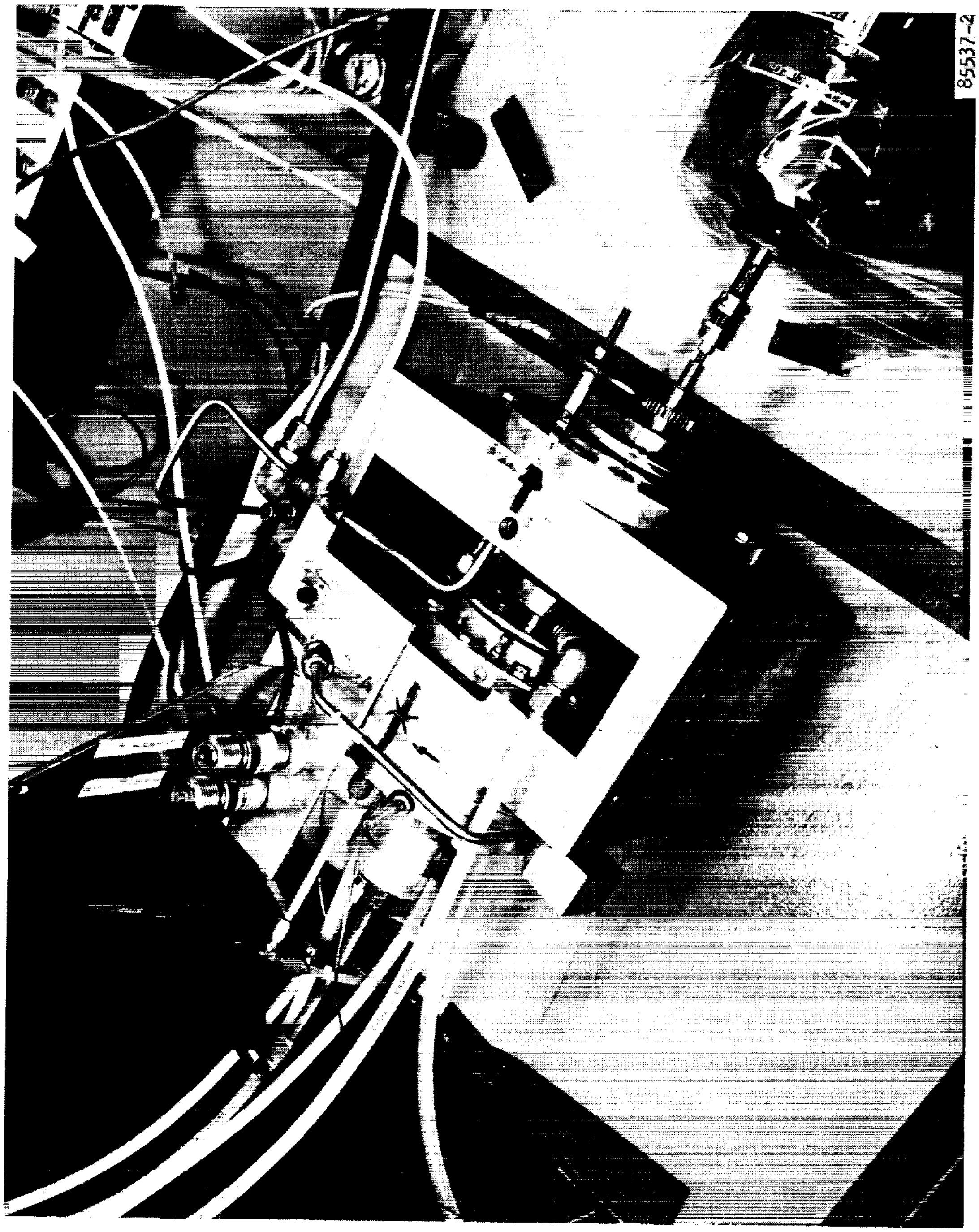

Figure 2-3. Test Setup Showing Closeup of Dynamometer

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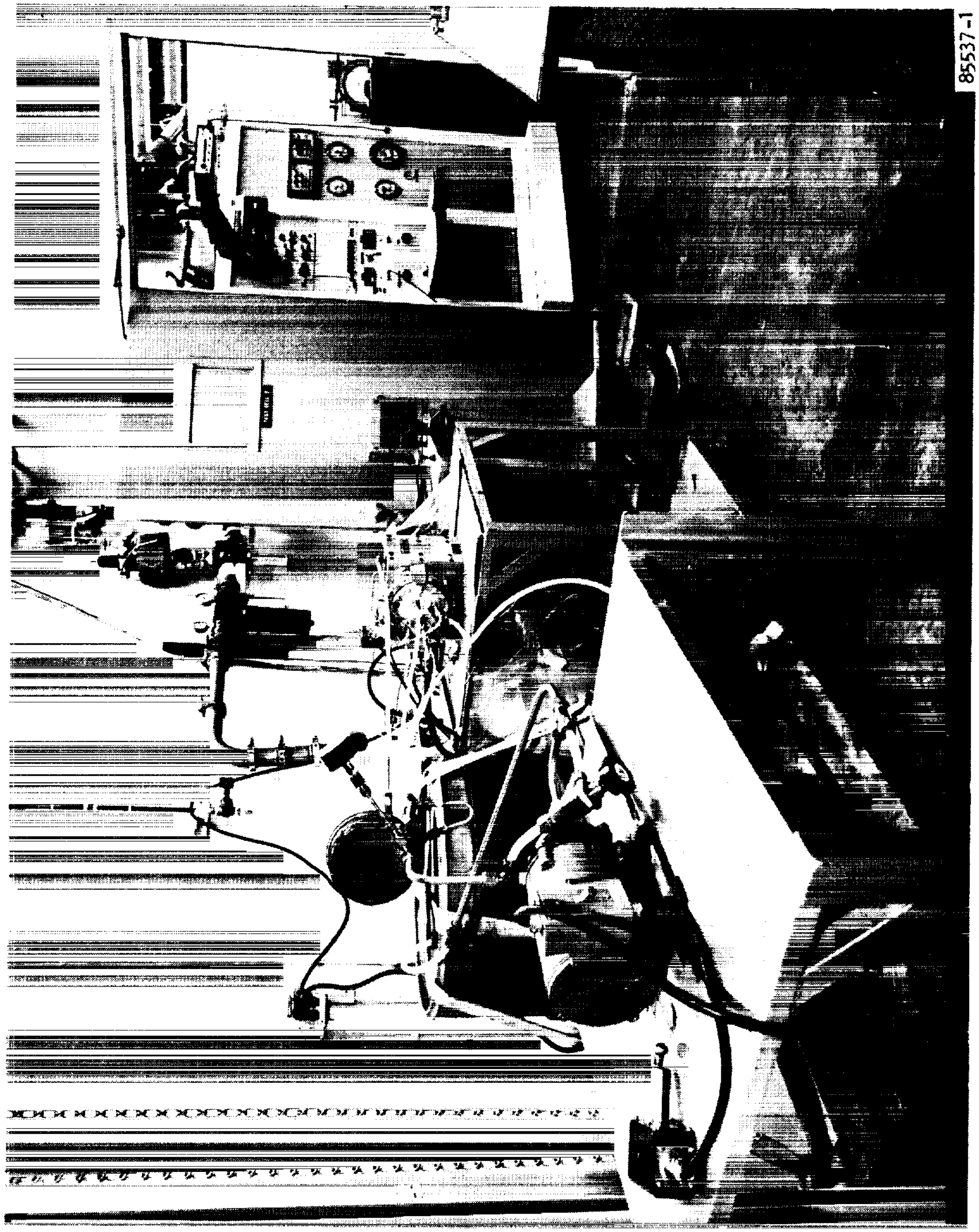

Figure 2-4. Overall Test Setup

3. TEST RESULTS

The initial tests began with several starts and stops to determine full-film speed and wear characteristics. Full-film speed is defined as the speed at which the bearing torque drops to a minimum value. This speed was approximately 200 rpm at a radial load of 4 lb. After approximately 20 starts and stops, the bearing showed no signs of wear. The bearings were therefore reinstalled and used throughout the remainder of the tests.

Tests to determine the load capacity were done by incrementally increasing the applied load and monitoring the torque. The maximum load capacity condition was usually indicated by a sudden increase in running torque, which was created by a rub between the shaft and the foil segments. Test speed ranged from 500 to 1750 rpm. Based on the water-to-LH_2 viscosity ratio at the normal pump operating condition (at 50°R and 450 psia), a 600-rpm test speed in water corresponds approximately to the LH_2 pump operating speed of 50,000 rpm. Maximum bearing load capacity at speeds over 1000 rpm were limited by the capacity of the pneumatic loading device.

The measured running torque vs applied load are shown in Figures 3-1 to 3-7 for various test speeds. Figure 3-2 shows that the load capacity at 600 rpm is 140 lb (33-psi bearing specific load). The largest load attained is 190 lb (45-psi specific load) at 1000 rpm.

The 33-psi load capacity at 600 rpm in water well exceeds the predicted minimum load capacity of 15 psi for the 1.75-in.-dia bearing. Furthermore, due to high speed and low viscosity, the turbulence effect at 50,000 rpm in an LH_2 pump will further enhance the load capacity of the bearing. During Phase I it was determined that the rotor weight, hydraulic radial unbalance load, and gyroscopic maneuver load of the preliminary pump required a bearing load of only 4 psi.*

Shaft dynamic runout was monitored at each speed and load. Higher loads tend to slightly decrease the runout amplitude, whereas higher speeds tend to slightly increase the runout amplitude. From the runout peak-to-peak amplitudes listed in Table 3-1, it is seen that the typical runout is 0.001 in. or less, which is very good.

After completing the load capacity tests, including more than 50 starts and stops, the bearing was removed and examined for wear. A photograph of the loaded section of the bearing after the water tests is shown in Figure 3-8. The wear was negligible, with much less wear than a comparable test in air. This is primarily due to the lower full-film speed in water (i.e., higher viscosity). Wear takes place only during lift-off, when the shaft speed is less than the full-film speed. As fluid viscosity increases, the full-film speed is reduced; this in turn reduces wear.

*See Phase I Interim Report, AiResearch Report No. 80-16884.

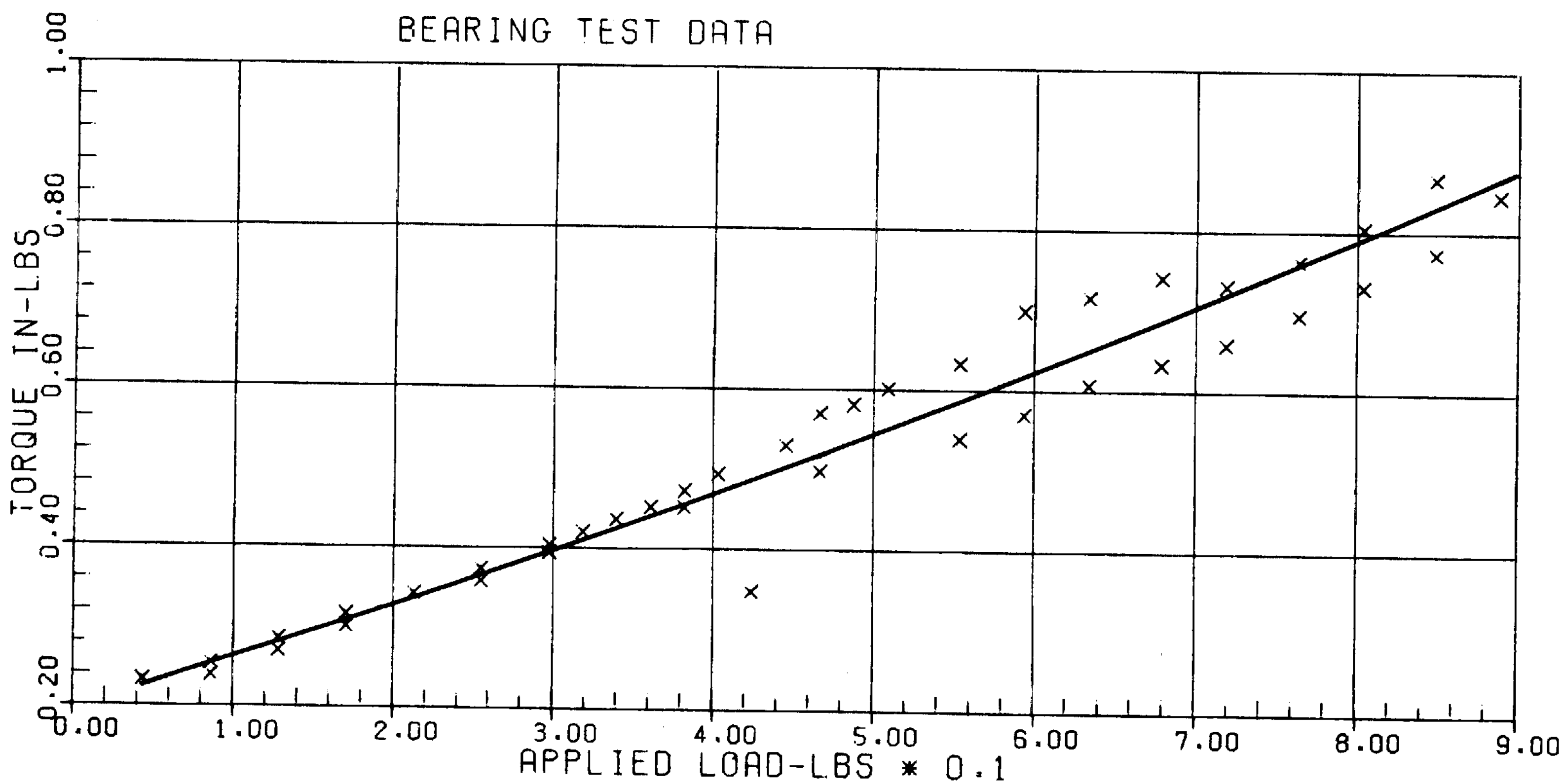

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Figure 3-1. Running Torque vs Speed at 500 rpm

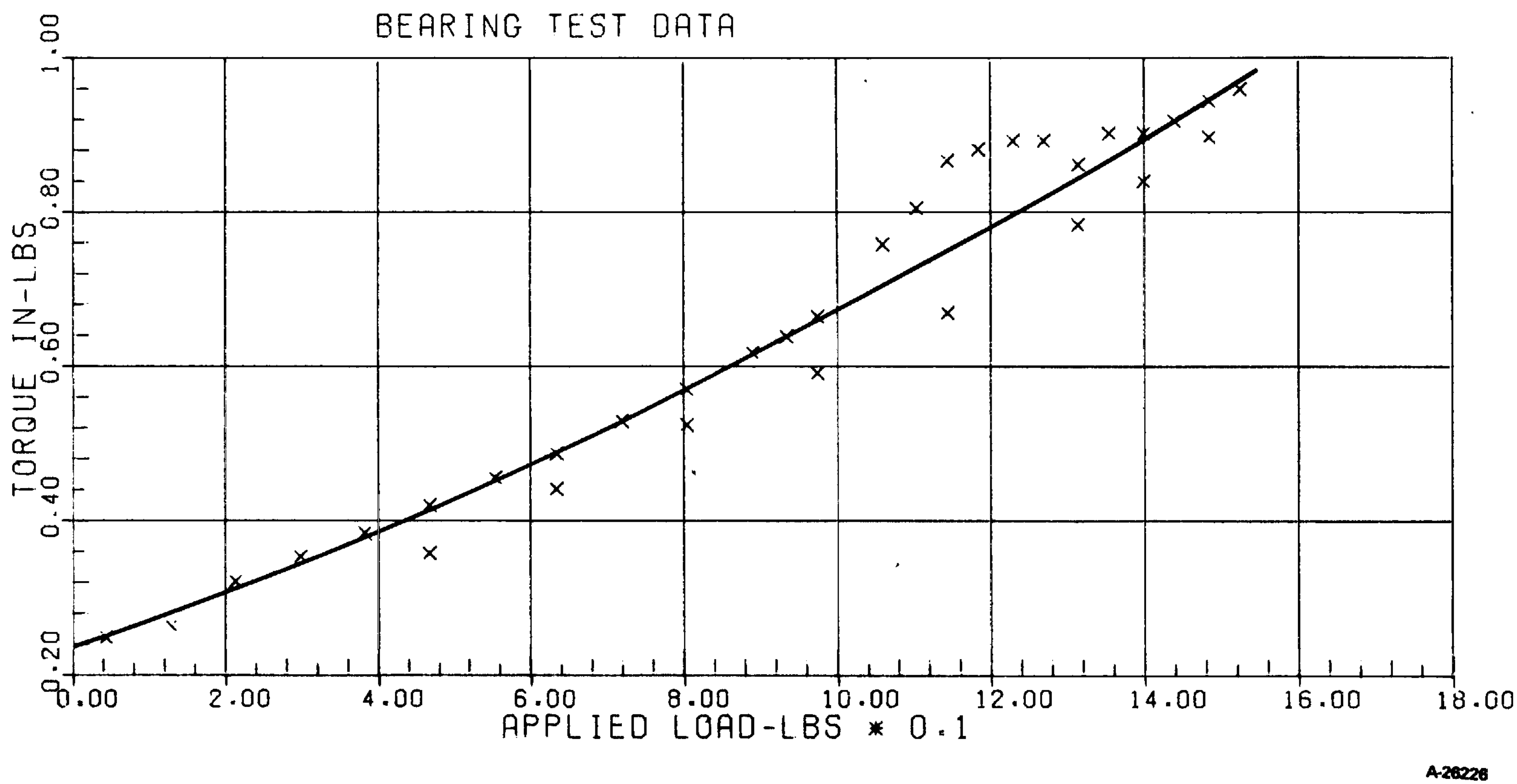

Figure 3-2. Running Torque vs Speed at 600 rpm

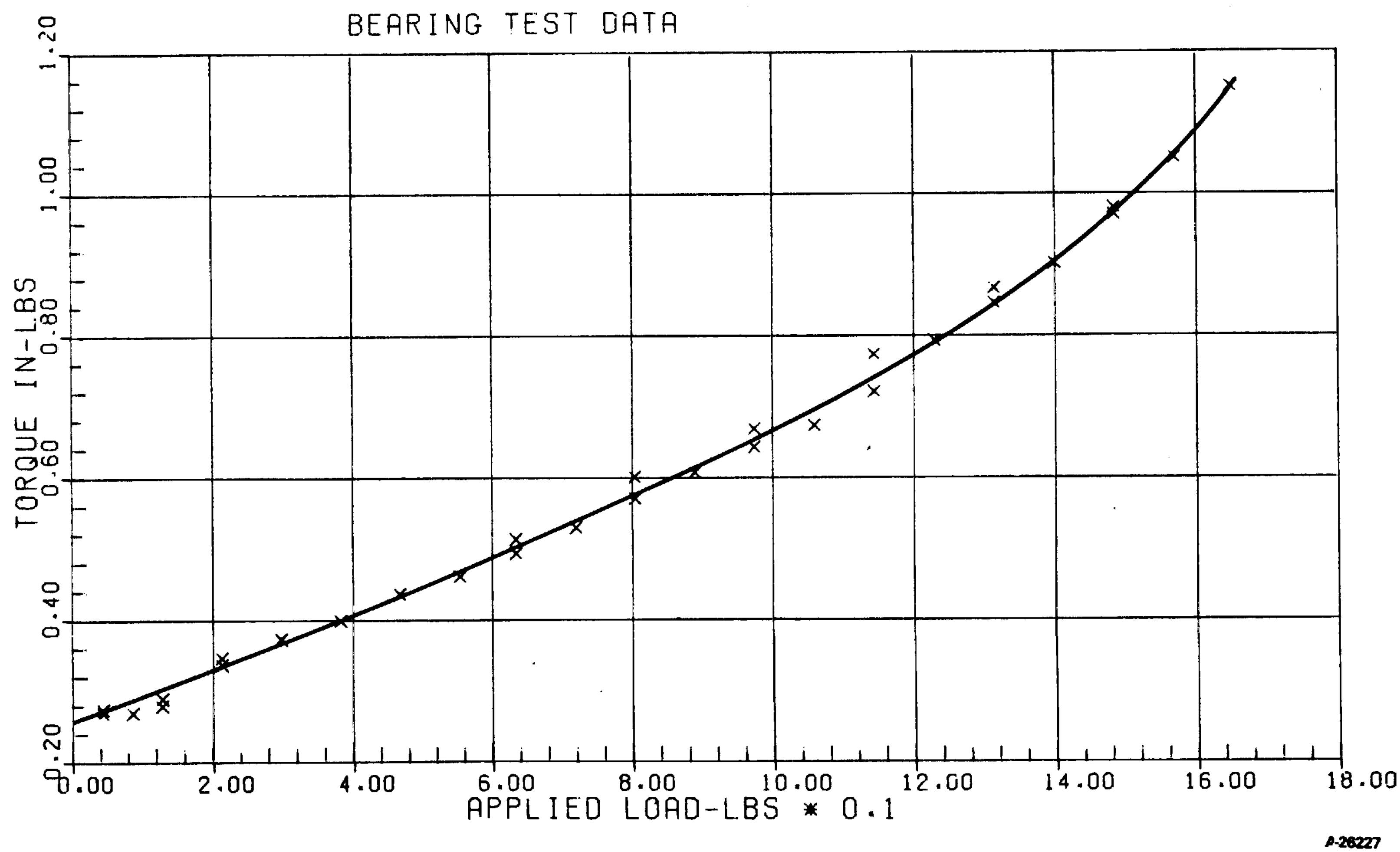

Figure 3-3. Running Torque vs Speed at 750 rpm

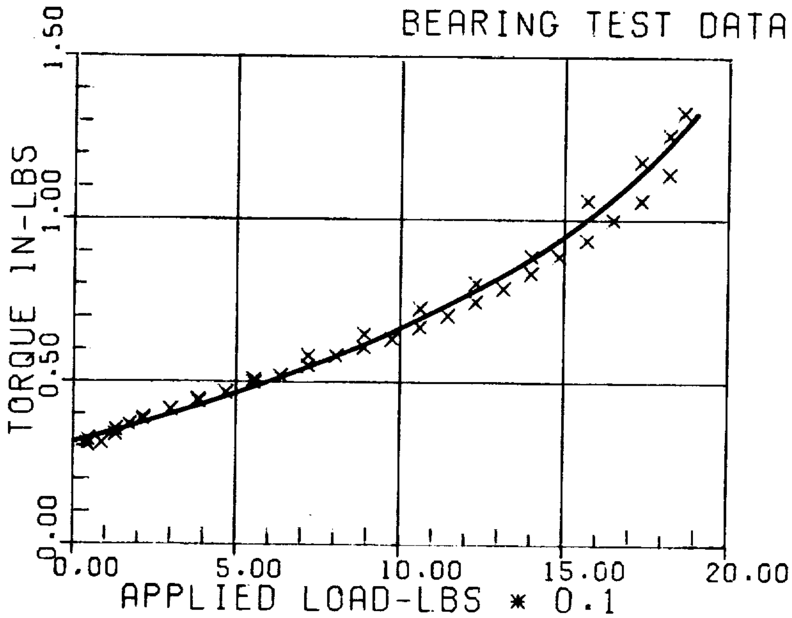

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Figure 3-4. Running Torque vs Speed at 1000 rpm

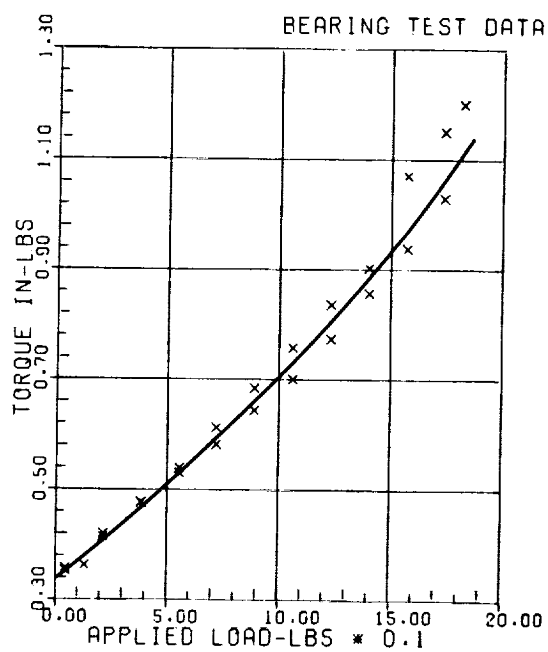

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Figure 3-5. Running Torque vs Speed at 1250 rpm

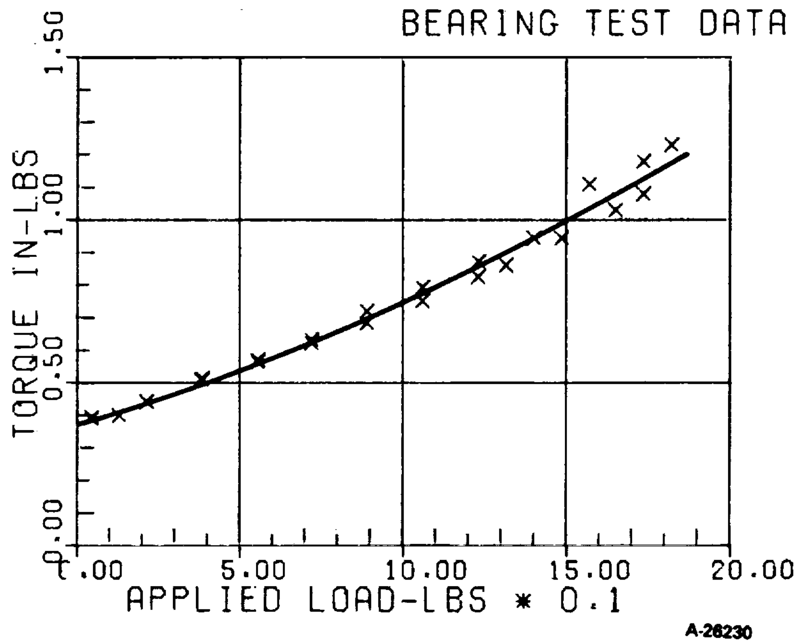

Figure 3-6. Running Torque vs Speed at 1500 rpm

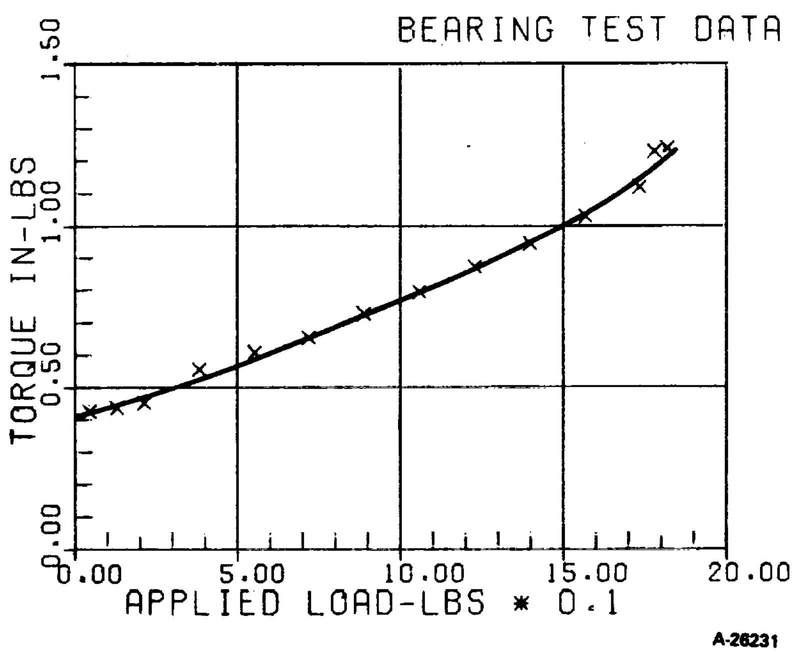

Figure 3-7. Running Torque vs Speed at 1750 rpm

TABLE 3-1

SHAFT RUNOUT

	Speed, rpm					
	600	600	1250	1250	1750	1750
Load, lb	4	144	14	169	4	169
Peak-to-peak runout, in.	0.0009	0.0006	0.0010	0.0007	0.0010	0.0008

At room temperature, the viscosity of LH_2 is 56 percent of air viscosity. The full-film speed and thus the rubbing velocity in LH_2 is 1.8 times that of the air. All bearing data indicate a very small coating wear rate in air. Even with a greater full-film speed, the coating wear rate is probably adequate in LH_2 pumps. However, since the wear coefficient of the bearing coating in an LH_2 environment is unknown, the question of wear rate or coating life can not be fully assessed at this time.

The measured full-film speed during the water tests was about 200 rpm; based on the viscosity ratio, this yields an equivalent full-film speed of 16,700 rpm in an LH2 pump. This value agrees with the full-film speed of 17,000 rpm predicted in the Phase I report (Figure 2-7 therein). The measured combined surface roughness for the shaft-end foil coating was 15×10^{-6} in.

At the pump equivalent speed of 600 rpm, the running torque from Figure 3-2 is 0.028 in.-lb at a bearing load of 16 lb, which is the total effective bearing load calculated in Phase I. Using this running torque value, the bearing power loss in an LH_2 pump, based on the laminar flow assumption, is 164 w. If the turbulence correction factor employed in the Phase I report were used, the turbulence effects in the LH2 pump would increase the power loss to about 260 w.

This power loss is about double the predicted value. This discrepancy could be partly due to the error in the torque measurement--an error that resulted from the interaction of water and air in the supporting dynamometer (see Figure 2-1), and from the uncertainty in the turbulence computation.

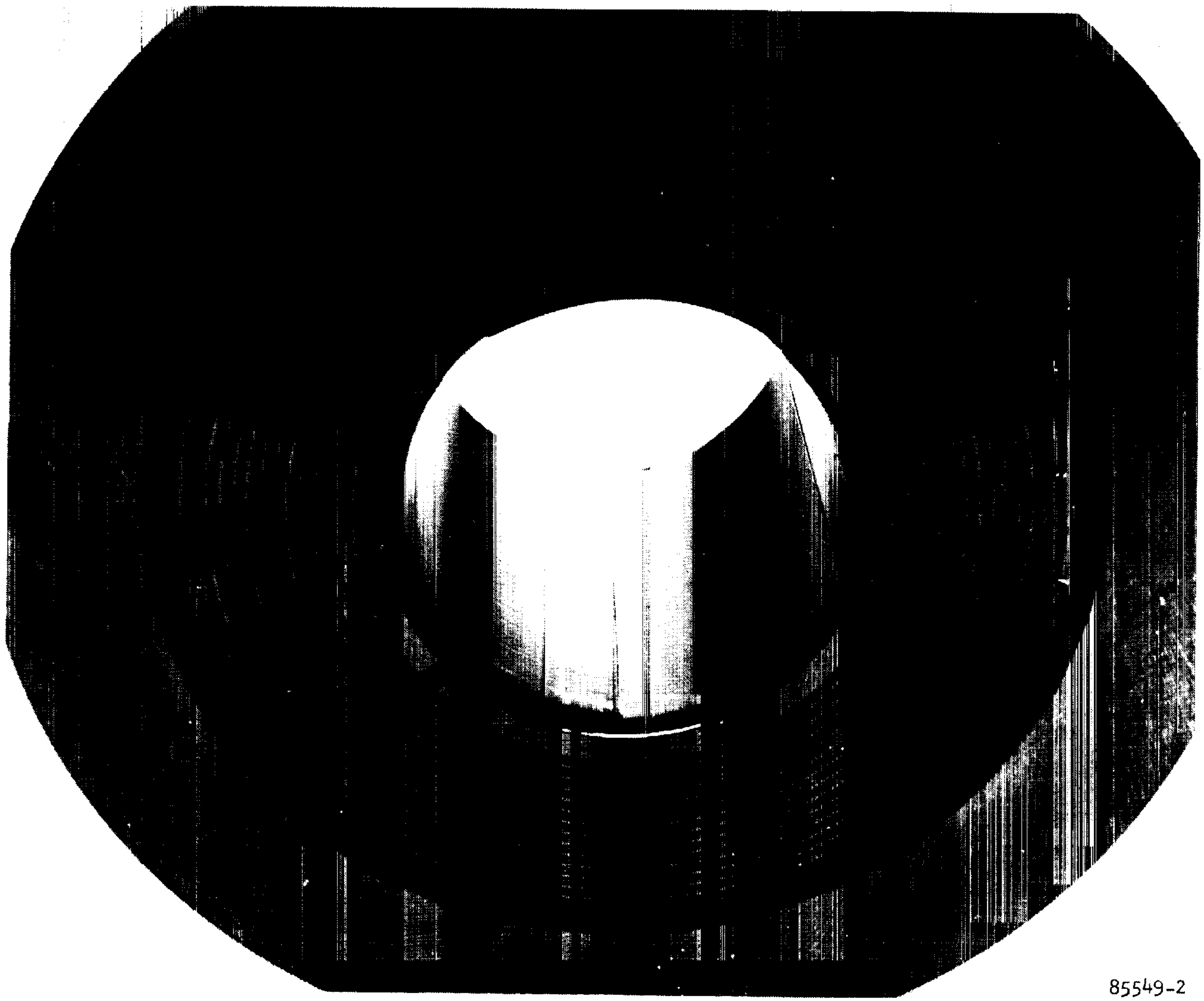

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Figure 3-8. Loaded Section of Bearing After Tests

4. CONCLUSIONS AND RECOMMENDATIONS

Major conclusions are summarized below:

(a) The selected test bearing size of 1.75-in. dia by 2.4 in. long demonstrated ample load capacity to support the LH_2 pump rotor. The measured maximum load capacity at 1000 rpm in water is 45 psi.

(b) The shaft dynamics in terms of dynamic runout were very good for the entire speed range of 500 to 1750 rpm. The same bearing has been tested in air to 75,000 rpm and showed good stability.

(c) The measured full-film speed agrees very well with the predicted value. In an LH_2 pump the full-film speed is estimated to be 16,700 rpm.

(d) Limited start-and-stop tests show negligible bearing wear in water.

(e) Measured power loss is higher than predicted. However, the level of power loss (about 260 w in an LH_2 pump) should be adequate.

(f) Compliant foil bearings appear very feasible for long-life LH2 pump applications. A bearing smaller than the tested bearing is probably adequate for the baseline preliminary pump design described in NASA CR-14539.

As mentioned earlier the wear rate of the Teflon coating in an LH_2 environment is unknown. Although no wear problem was experienced in the water tests, a simple and direct wear test in LH2 is recommended. A simple wear rig shown in Figure 4-1 could be used. A commercial electric motor would be used to drive an existing vertically mounted test shaft. The shaft would be supported by the foil bearing on one end (submerged in 1 atm LH2), and by a ball bearing on the other. Wear tests would be run by cycling the speed from 0 to 16,700 rpm, which is the bearing full-film speed. The number of cycles could be determined from the start-and-stop cycle requirement for LH_2 pumps.

After the wear test, total feasibility of foil bearing application in long-life LH_2 pumps would be established.

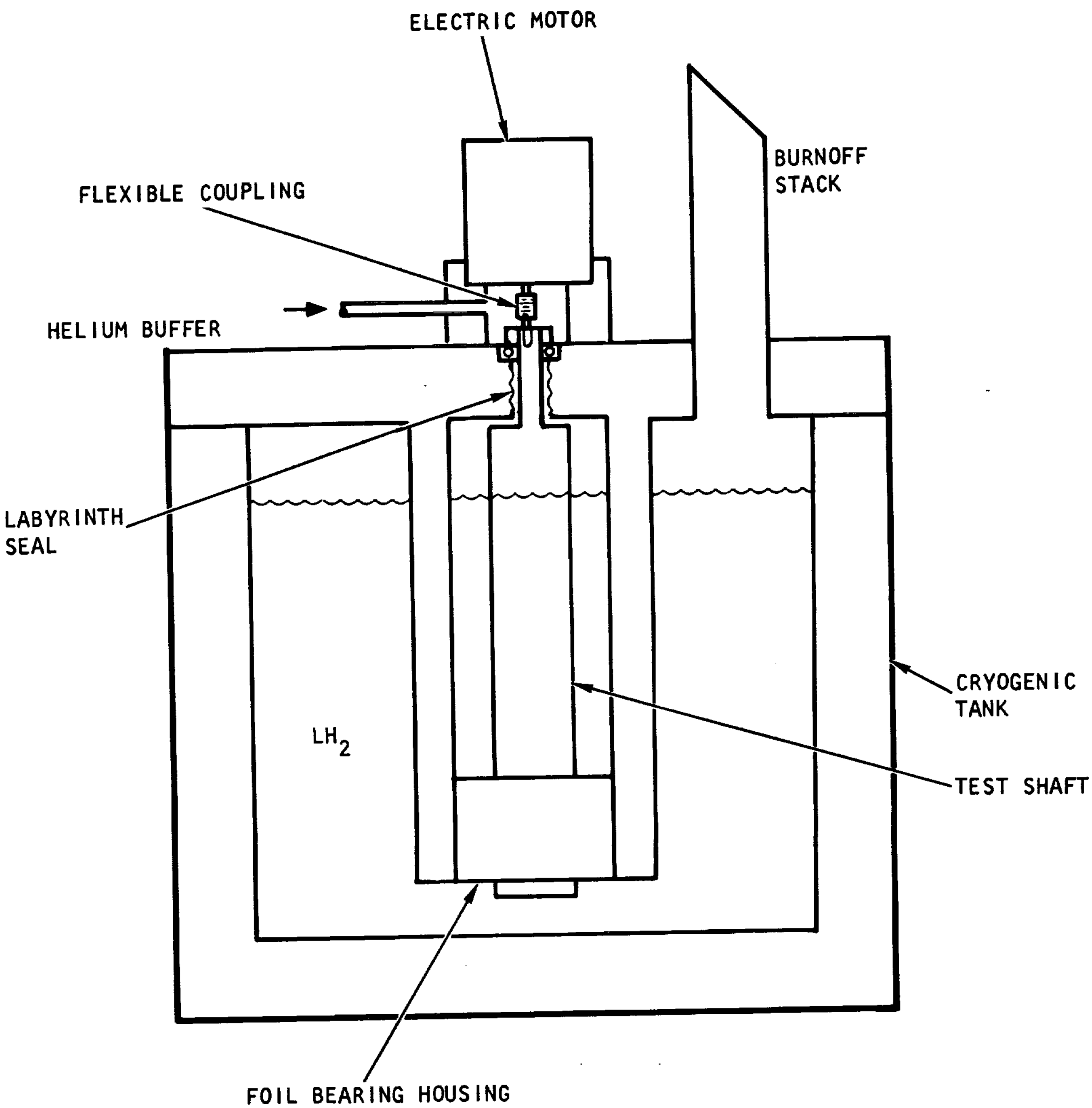

Figure 4-1. Wear Test Rig Schematic